THE KILDARE CORPORATION
One Spar Yard Road
New London, CT 06320-5527
U.S.A.

Tel.(860) 443-7768
e-mail: kildare@connix.com

FAX (860)437-4587

TKC 8.31

"SYNTHETIC-BLUBBER" COMPLIANT COATINGS

For

REDUCTION of

HYDRODYNAMIC-DRAG and FLOW-NOISE

Based on

THE MATCHED SHEAR IMPEDANCE HYPOTHESIS

By:
James W. Fitzgerald
Edwin R. Fitzgerald
James E. Martin
Eugene F. Modert

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James W. Fitzgerald, James E. Martin
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1 Spar Yard Street
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13. ABSTRACT After some 40 years of extensive investigations, the issue of whether or not compliant surfaces can reduce hydrodynamic drag remains unresolved. Based on Kramer's initial identification of the dolphin's skin as the basic mechanism, most of the past work has concentrated on thin (~ 0.3) compliant coatings, presumed to deflect in some preferred manner and interact with the boundary layer Tollmeir-Schlicting waves so as to delay the transition from laminar to turbulent flow. The Matched Shear Impedance Hypothesis takes a completely different approach more akin to "acoustica" than "hydrodynamics". Identifying the thick (~3 cm) blubber as the effective compliant "layer", the basic postulates are:

- The Turbulent Boundary Layer (TBL) is viewed as a fluctuating "Shear Stress Generator" coupled to the "Compliant Layer Load" through the viscous inner boundary layer.
- To transfer appreciable energy (power) from the TBL Shear Stress Generator to the Compliant Layer Load, the Shear Impedance of the Load, must be "matched" to the Shear Impedance of the TBL Generator.
- Under Matched-Load conditions, the build-up of the fluctuating energy of incipient turbulence is reduced by energy flow into the compliant layer where it is dissipated by losses in the viscoelastic compliant layer material, thus delaying the onset of turbulence.
- Dolphin blubber represents just such a matched-load with the required high loss tangent.

Preliminary Rotating Disc measurements of drag (torque) on a polymethyl siloxane gel/polyurethane (PMS) composite sample delayed the onset of turbulence to $\omega \cong 260$ rpm as compared to $\omega \cong 210$ rpm for the rigid reference disc. The PMS sample was not fully matched to blubber and has the same surface roughness. The projection of a smooth and fully matched sample indicates a potential laminar-to-turbulent transition at $\omega \cong 448$ rpm, corresponding to peripheral velocity of ~ 10 knots and a Reynold's No. $\cong 4.7 \times 10^5$.

14. SUBJECT TERMS

- Compliant Surfaces
- Synthetic Blubber
- Hydrodynamic Drag
- Flow Noise
- Matched Shear Impedance Hypothesis
- Complex Dynamic Shear Compliance

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1.0 INTRODUCTION

1.1 Program Identification

This report constitutes data item A002, Final Technical Report, of the contract Data Requirements List for Contract No. DAAH01-95-C-R013, and represents work done under Phase-II of the following DARPA SBIR:

DARPA 93-032 TITLE: Synthetic Dolphin Blubber Compliant Coatings

CATEGORY: 6.2 Exploratory Development

OBJECTIVE: Develop synthetic materials based on the viscoelastic properties of live dolphin blubber to act as a compliant coating for underwater vehicles to reduce drag as well as noise. No experimentation with live dolphins will be conducted as part of this effort.

DESCRIPTION: The dolphin has a remarkable skin and blubber which minimizes both drag and flow noise. Recent studies of fast swimming dolphins in sea water show little phosphorescent activity due to the reduced turbulent boundary layer. Recent materials investigations have shown the unusual properties of the blubber. The purpose of this task is to synthesize the material so it will exhibit properties similar to those of live dolphin blubber.

Phase I: Develop, fabricate, and test synthetic blubber materials. Examine candidate coatings and measure viscoelastic properties. Three coatings will be selected for full-scale tests.

Phase II: Fabricate, install, and test the coatings developed in Phase I on undersea vehicles approximately 36 feet in length and 44 inches in diameter.

An undersea vehicle was not available for the Phase-II studies, nor would it have been within the budget constraints. Candidate "synthetic blubber" compositions were tested for drag reduction by means of a Rotating Disc Apparatus, as reported hereinafter.

1.2 Phase-I Program Results [1]

The frequency dependence from 2 to 1000 Hz of the complex shear compliance ($J^* = J' - iJ''$), or shear impedance ($G^* = G' + iG''$), were measured for 10 synthetic materials. Preliminarily, 3 polymer systems were identified as potential candidates compliant coatings with viscoelastic properties comparable with previous measurements on blubber. These were:

- Polyvinyl Chloride-Di-2-Ethylhexyl Phthalate (PVC-DOP)
- Polydimethyl Siloxane (PDM)
- Polyvinyl Chloride-Dimethyl Thianthrene (PVC-DMT)

For applications as coatings on vehicles, these candidate systems are considered to be too "tender" . . . as, indeed, would be the case for blubber, itself. The tear and penetration resistance is not sufficient for the operational environment of, for example, AUVs. Consequently, in Phase II composite systems were targeted, including: (a) sheathed coatings (e.g. a thin tough elastomer "skin" bonded to the "matched" gel coatings); (b) filled matrix systems (e.g. open-cell sponge filled with "matched" gels); and (c) a combination of (a) and (b).

A number of commercially available damping materials were measured but were found to be too stiff to match the driving point impedance of blubber. These included Sorbothane and several synthetic and natural rubber stocks.

An important adjunct to Phase II of this SBIR program would be access to Rotating Disk Apparatus in order to evaluate the actual drag-reduction performance of candidate compliant coatings before application to "test-bed" AUV. This apparatus was subsequently developed by KILDARE under a separate contract, but only became available for this Phase-II effort essentially after the nominal termination of this Contract. Subsequent measurements with the Rotating Disc Apparatus have delayed the completion of this Final Report.

1.3 Background

After some 40 years of extensive investigations, the issue of whether or not compliant surfaces can reduce hydrodynamic drag remains unresolved [2]. Based on Kramer's [3] initial identification of the dolphin's skin as the basic mechanism, most of the past work has concentrated on thin (~ 0.3 cm) compliant coatings. The various proposed hydrodynamic models [4,5] suggest that the compliant surface deflects in some preferred manner and interacts with the boundary layer Tollmein-Schlicting waves so as to reduce their stability, thereby delaying the transition from laminar to turbulent flow. Surprisingly, in spite of the extensive past work by many investigators, in most cases the compliant surface being studied has not been adequately characterized by measurement of its dynamic complex viscoelastic properties.

Most of the preceding investigations of hydrodynamic flow over compliant surfaces have been deficient in one or more of the following areas:

- Concentration on the thin skin of the dolphin rather than the thick blubber.
- Not characterizing the compliance of the surface by measurements of the dynamic complex shear compliance ($J^* = J' - iJ''$) and, hence, the driving-point shear impedance.
- Lack of a convenient laboratory method of measuring the drag, with sufficient precision and under controlled conditions.

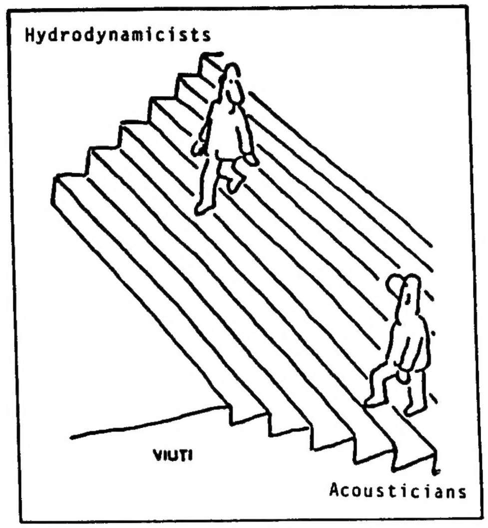
Our studies address each of these deficiencies.

1.4 The Matched Shear Impedance Hypothesis

The "Matched Shear Impedance Hypothesis for Compliant Layer Control of Boundary Layer Turbulence" takes a completely different approach to the problem than the normal hydrodynamic treatment . . . more akin to "acoustics" than "hydrodynamics" (Figure-1). This model's [6,7] basic postulates are:

THE
MATCHED
SHEAR
IMPEDANCE
HYPOTHESIS
FOR . . . .

COMPLIANT
LAYER
CONTROL OF
BOUNDARY
LAYER
TURBULENCE

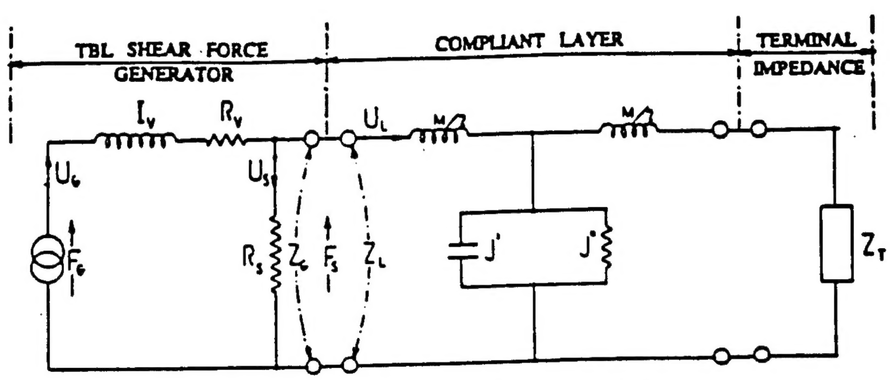


Not everything is the same to everybody

(Non omnia eadem aegue omnibvus)

Plautus, *Asinaria* (c 200BC)

Figure-1 Hydrodynamics vs. Acoustics



F_G = TBL Shear Force Generator
I_V = Rotational Inertia of Vortices

R_V = Viscous Losses in Fluid

R_S = Viscous Losses in Boundary Sub-layer
M = Unit Mass of Compliant Layer

J' = Elastic Compliance of Compliant Layer
J" = Loss Compliance of Compliant Layer

Z_T = Terminal Impedance
Z_G = Impedance "Looking Into" TBL (Generator Internal Impedance)
Z_L = Impedance "Looking Into" Compliant Layer (Load Impedance)
U_G = Fluctuating Component of Flow Velocity
U_S = Fluctuating Velocity Component in Boundary Sub-layer
U_L = Fluctuating Velocity in Compliant Layer
F_S = Shear Force Applied to Compliant Layer

Figure-2 Equivalent Circuit

- The Turbulent Boundary Layer (TBL) is viewed as a fluctuating "Shear Stress Generator" coupled to the "Compliant Layer Load" through the viscous inner boundary layer.
- To transfer appreciable energy (power) from the TBL Shear Stress Generator to the Compliant Layer Load, the Shear Impedance of the Load must be "matched" to the Shear Impedance of the TBL Generator.
- Under Matched-Load conditions, the build-up of the fluctuating energy of incipient turbulence is reduced by energy flow into the compliant layer where it is dissipated by losses in the viscoelastic compliant layer material, thus delaying the onset of turbulence.
- Dolphin blubber represents just such a matched-load with the required high loss tangent.

Figure-2 shows an Equivalent Circuit representation of the Matched Shear Impedance Hypothesis.

2.0 MEASUREMENT METHODS

2.1 Viscoelastic Measurement Method

An automated dynamic mechanical system for complex shear compliance, $J^* = J' - iJ''$, and shear modulus, $G^* = G' + iG'' = 1/J^*$, loss tangent, $J''/J' = G''/G'$ and shear wave velocity and attenuation was used to get these parameters for dolphin blubber and the compliant coating materials. The system is one in which the complex mechanical impedance (force/velocity) of a rigid plate, with fine wire embedded coils suspended transversely in permanent magnet fields, is obtained in terms of a measured transfer electrical admittance (current/emf). From calibration values of the mechanical impedance of the plate alone, Z_{mp}, and an increased impedance Z^*_{mt}, with a pair of thin disc-shaped samples pressed against the plate, the sample impedance by subtraction is,

$$Z^*_{ms} = Z^*_{mt} - Z^*_{mp} \quad . . . . (1)$$

for samples of cross sectional area, A, and thickness, h, the complex shear compliance is then,

$$J^* = (-iY^*_{ms} A/h / 2\pi f) \quad . . . . (2)$$

where $Y^*_{ms} = 1/Z^*_{ms}$ and f = the vibration frequency. The elastic component, J', represents the in-phase component of shear strain/stress, while the viscous J'' component, J'', is delayed, 90° out of phase strain/stress. This system has a frequency range from 2 to 10,000 Hz at temperatures from - 50 to 150° C. A complete description is given in several publications [8,9].

2.2 Rotating Disc Apparatus

Pioneering work in utilizing rotating disc apparatus for studying flow over compliant surfaces was done by Hansen & Hunston, [10]. Their apparatus used thin discs, ~ 0.98 cm

thick by ~ 20.9 cm diameter, and covered a range of Re ≅ 10^4 to Re ≅ 5×10^5. The compliant coatings studied were thin (~ 0.34 cm) soft plastisols ($|J^*| \cong 3 \times 10^{-4}$ cm^2/dyne) that developed surface instabilities, accompanied by a marked increase in torque (drag) in the vicinity of Re ≅ 5×10^4. Another set of rotating disc experiments were made by Chung and Merrill, [11] on (thin) coatings of a (soft) silicone rubber with a diluent silicone oil. A pronounced rippling of the surface was accompanied by marked increase in drag in the vicinity of Re ≅ 10^4.

The isometric sketch of Figure-3 shows the Rotating Disc Apparatus available for this study. It consists of a variable speed motor connected, by means of a pulley-belt drive, to a shaft mounted disc, rotating in a water-bath. The drive-shaft has an in-line torque/rpm sensor whose output are read by a digital meters. The equipment measures both torque (0-100 lbs-in) and rotational speed (0 - 10,000 rpm). Both analogue (± 5 volt) and digital (RS-252-C) outputs are available, in addition to the panel meters.

Figure-4 shows the design of the rotating discs. The basic rigid (aluminum) reference disc is 20 cm in diameter and 4.5 cm thick, with edges having a radius of ~ 0.16 cm. The molded compliant layered discs have the same outside dimensions and surface-smoothness as the rigid-reference discs; but, three different thicknesses of compliant layers: viz. 0.5 cm, 1.0 cm, and 2.0 cm. This means that, for a given compliant material, there will be a 4 to 1 range in the "static" shear deflection driving-point shear impedance. The effect of the surface compliance (i.e., the driving point impedance), if any, will be the difference between the torque of the rigid-reference disc and that of the compliant-layered disc. Significant differences in drag (torque) as small as a few percent can be determined.

In summary, the Rotating Disc Apparatus has the Apparatus has the following measurement capabilities:

- Rotating speed range; 60 rpm to 2250 rpm
- Torque range; 0 - 100 lb-in
- Compliant layer thickness; 0.5, 1.0, & 2.0 cm
- Rotating discs, 4.5 cm thick x 20 cm diameter
- Reynold's number range; 5×10^4 to 2.25×10^6
- Drag (torque) measurement precision; ~ ± 0.5%
- Measurements at room temperature, only

2.3 Rotating Disc Apparatus Calibration

One of the problems resulting from the required thickness of the rotating discs, particularly at the higher rotational speeds, is that the rotating disc "stirs" the bath (30" D x 24" H) into a general rotating water mass. the relative velocity of the rotating discs through the water is reduced, accompanied by a reduction in torque (drag). The Himmelstein Precision Torque Meter Readout (Model 66042) and In Line Torque Sensor (Model MCRT 2901 T) has an A/D conversion time of only 30 microseconds. Stability is achieved after one second, the same time base upon which the instrument output is gated. Sampling occurs as an integration of the output over each second. Measurements of torque vs time indicated that even at the highest rotational speeds, the torque

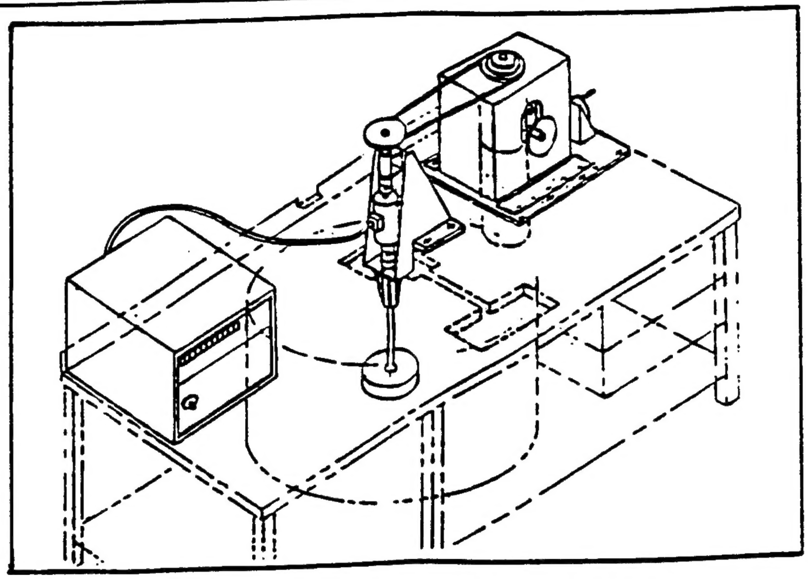


Figure-3 Rotating Disc Apparatus

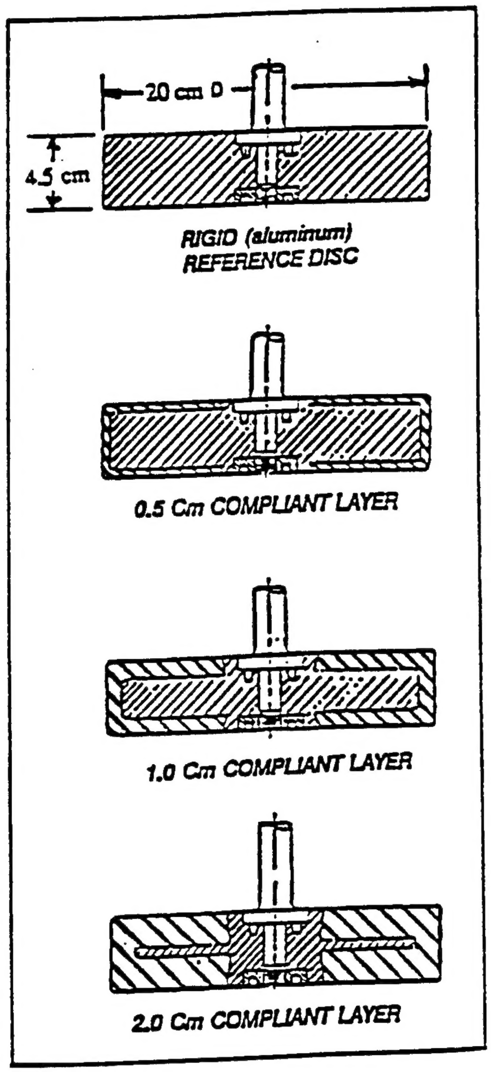


Figure-4 Rotating Discs

remained substantially constant in a 2 - 4 second window and then begins to decay as the water mass rotation sets in. In effect, the disc, driven by the 2 hp electric motor, reaches it terminal rotating speed in ~ 1 second. Measurements made in the 2 - 4 second period following, represents the true rotational speed and torque through quiescent water. We, therefore, adopted the "2nd-second" measurement method throughout our studies.

Table-I shows a series of torque measurements made on the thick rigid-reference disc, using the "2nd-second" method. It should be noted that these are completely independent measurements starting with a non-rotating disc in quiescent water. The mean of the 10 runs was T = 19.33 lb-in, with a standard deviation of $\sigma = 0.068$. The maximum spread was only 0.20 lb-in, or ~ ± 0.5%. This represents a measurement precision not often encountered in hydrodynamic drag measurements.

Table-I: Repeatability

RPM	TORQUE (lb.-in)	STATS
1000	19.3	19.33 Mean
1000	19.2	0.0675 Std Dev
1000	19.3	19.4 Max
1000	19.3	19.2 Min
1000	19.4	0.200 Spread
1000	19.3	
1000	19.4	
1000	19.4	
1000	19.3	
1000	19.4	

2.4 Torque Coefficient

The customary dimensionless torque coefficient is:

$$C_m = \frac{T}{1/2\rho\ \omega^2 R^5}\ ; \qquad (1)$$

and Reynold's number for the rotating disc is:

$$Re = \frac{(R^2)(\omega)}{\upsilon} \qquad (2)$$

where:

C_m = torque coefficient
T = measured torque
r = density of water
ω = rotational speed
R = disc radius
Re = Reynold's number
υ = kinematic viscosity of water

Figure-5 shows the torque vs. rpm calibration of the rigid disc. Transition from laminar to turbulent flow occurs at $\omega = 210$ rpm, which corresponds to a peripheral speed of $V \cong 4.3$ knots. Figure-6 shows a corresponding rigid disc drag coefficient vs. Reynolds No., with laminar-to-turbulent flow at $Re \cong 2.2 \times 10^5$.

In the laminar flow regime:

$$C_m \text{ (laminar)} \doteq (3.9 \times 10^6)(Re^{-1.4}), \qquad (3)$$

and in the turbulent regime:

$$C_m \text{ (turbulent)} \doteq (2.5 \times 10^3)(Re^{-0.8}) \qquad (4)$$

These numerical equations characterize the hydrodynamic drag (torque) on the rigid reference disc.

3.0 ROTATING DISC DRAG MEASUREMENTS

3.1 Representative Compliant Materials

Figure-7 shows the shear compliance $|J^*|$ vs. frequency for some representative compliant materials from a companion study [12]. The materials are identified as follows:

PDM - 10:1 polymercuring agent polydimethyl siloxane gel
BLB - dolphin blubber
PMS - polymethyl siloxane gel/polyurethane foam composite
15SILR - Shore A 15 durometer silicone rubber
35NEOR - Shore A 35 durometer Neoprene rubber
55NEOR - Shore A 55 durometer Neoprene rubber

3.2 Drag (Torque) Measurements

Rotating discs measurements on the Neoprene rubbers showed no perceptible drag differences from the drag measurements on the reference rigid discs. The molded 15-durometer silicone rubber discs also showed no perceptible delay of turbulence, but surface flaws may have masked any effect, and this sample will have to be remolded and rerun. No rotating disc samples have yet been made with the PDM (polydimethyl silicone gel), but with a complex shear module of $|J^*| = 2 \times 10^{-5}$ cm / dyne, surface deflections can be expected to increase the drag.

Figure-8 shows the comparison of the transition from laminar to turbulent regimes for the rigid reference disc and the PMS (polymer gel-foam) coated disc. The transition takes place at $\omega \cong 210$ rpm, corresponding to $Re \cong 2.2 \times 10^{-5}$, for the rigid disc. The transition is delayed to $\omega \cong 260$ rpm, corresponding to $Re \cong 2.7 \times 10^5$, for the PMS disc. These preliminary measurements were made with a gel-foam compliant layer not fully matched to blubber . . . $|J^*| \cong 10 \times 10^{-7}$ cm^2 / dyne for PMS as compared to $|J^*| \cong 20 \times 10^{-7}$ cm^2 / dyne for BLB. Moreover, the surface roughness of the gel-foam was excessive and sample preparation techniques will have to be improved.

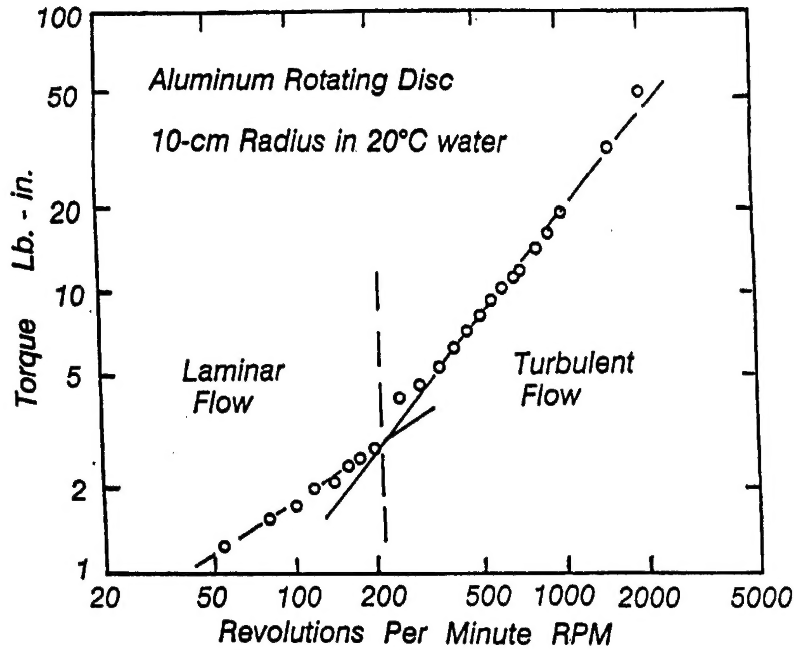
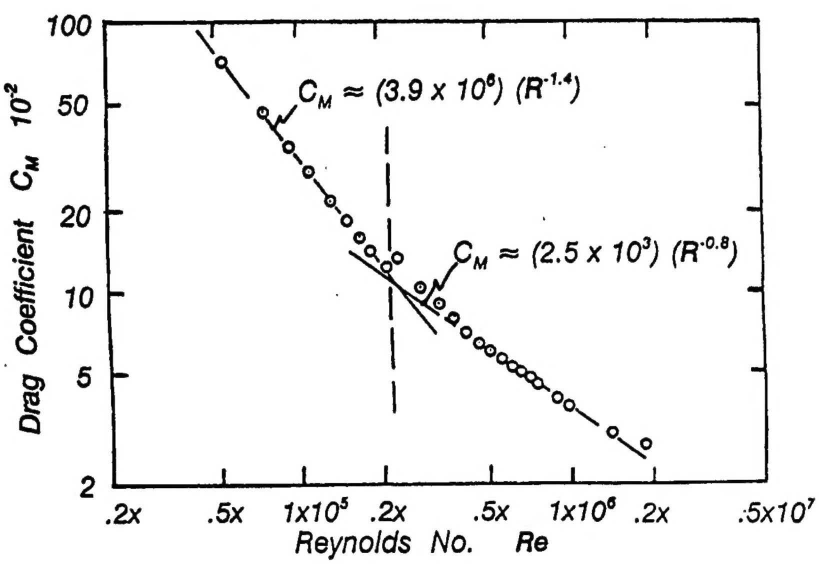


Figure-5 Rigid Disc Torque vs. rpm



$$C_m = \frac{T}{1/2 \rho\, \omega^2 R^5}$$

$$Re = \frac{R^2 \omega}{\nu}$$

Figure-6 Rigid Disc Drag Coefficient vs. Reynolds No.

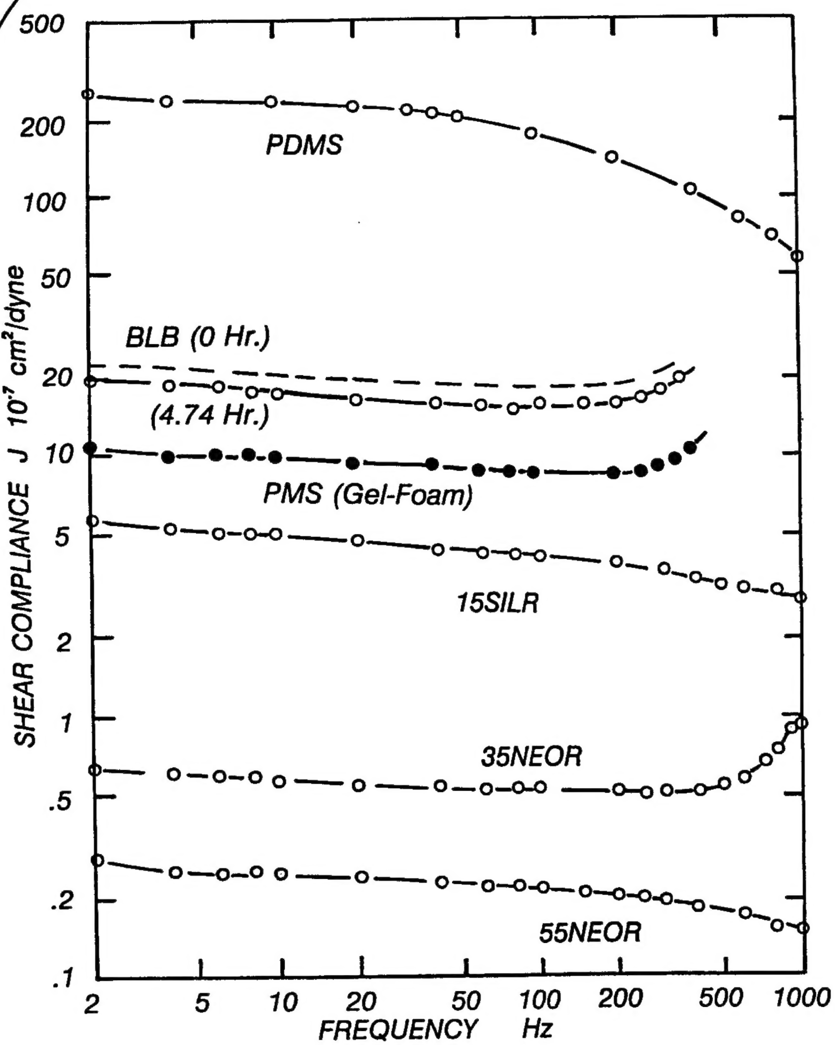


Figure-7 Shear Compliance of Representative Compliant Materials

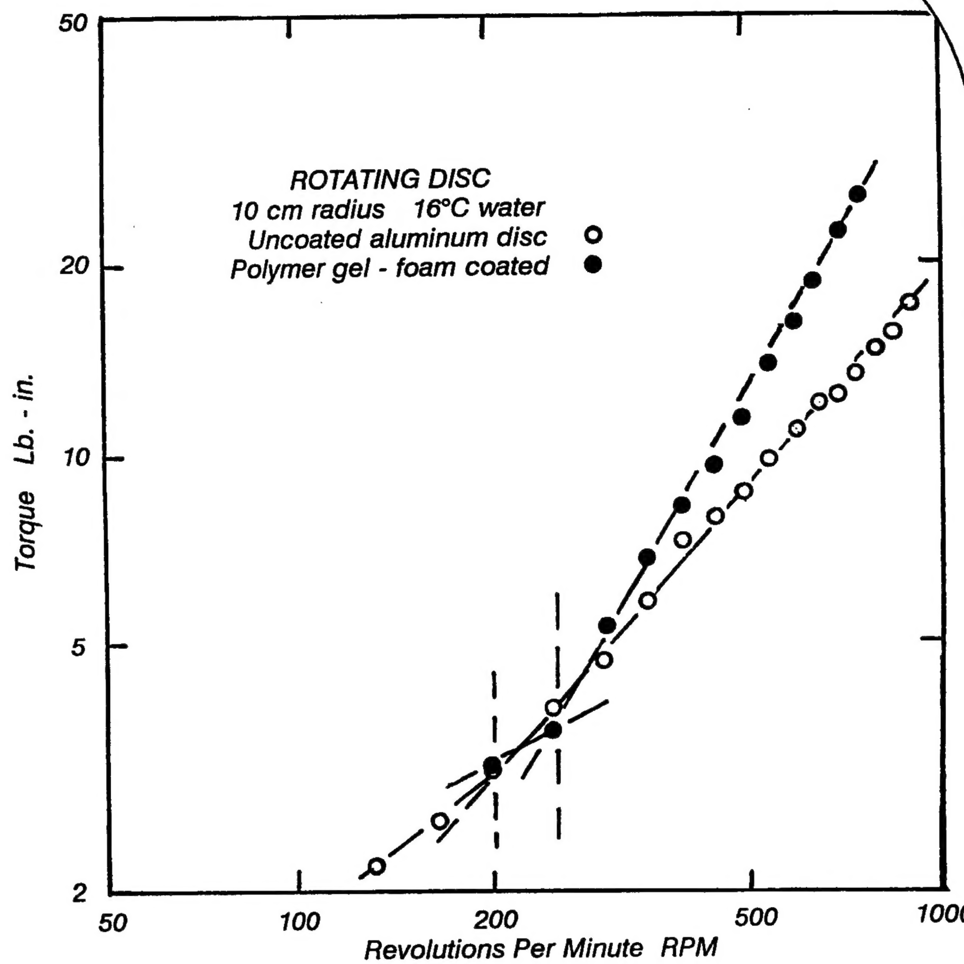


Figure-8 Comparison of the transition from laminar to turbulent regimes for the Rigid Reference Rotating and a Polymer Gel-Foam Coated Disc

4.0 CLOSING REMARKS

The gel-foam sample transition from the laminar to the turbulent regime at $\omega \cong 260$ rpm corresponds to a peripheral velocity of $v \cong 5.8$ knots. A fully matched, smooth sample could be expected to extend this delay of the onset of turbulence even further. Operational dolphins typically cruise at ~ 10 knots [13]. If we assume that this represents an "energy conserving" speed corresponding to the laminar-turbulent transition, a smooth fully matched compliant coated disc might be expected to extend the transition further to ~ 448 rpm, corresponding to the dolphin cruising speed of ~ 10 knots, or a Reynolds Number, $Re \cong 4.7 \times 10^5$.

These preliminary rotating disc measurements appear to support the "Matched Shear Impedance Hypothesis for Compliant Boundary Layer Control of Boundary Layer Turbulence"

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